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AN
INVESTIGATION
AND
DISCUSSION
OF
CYLINDER
CONDENSATION.

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The object of this thesis is an investigation of the heat changes in a steam engine, with an attempt to form some idea of what causes the phenomenon known as initial cylinder condensation and how it takes place. We know that the losses in an engine, due to this cause are very large and excessive. Possibly if we knew just what happened in the steam engine cylinder itself to cause this initial condensation, some means might be devised to prevent or reduce it and thus make a very great saving in the cost of steam power, the engine being by this means made

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to give much more economical results.

We propose to give the thesis under the following heads.

- 1- Outline of what was done, and the methods of taking the necessary readings. p. 4
- 2- Instruments used with their descriptions, constants, and calibrations. p. 8
- 3- Engine used in making the tests, with its constants. p. 34
- 4- Steam tables etc. used in working up results. p. 37
- 5- Calculations of test results;
I.H.P., B.H.P., eff., etc. p. 38
- 6- Diagrams, with their explanations and the methods used in making them. p. 46
- 7- Method employed to draw

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curves showing the variation of heat with time. p. 62

8- Curves between heat and time, with their explanations, etc. p. 67

9- Sources of error in these calculations and curves. p. 80

10- Possible theories and conclusions. p. 83

1- Tests were made on a steam engine, with readings taken to determine the following quantities: Steam consumption; revolutions per minute; quality and pressure of steam supplied; quality and pressure of the steam in the

exhaust. I incidentally to this, readings were taken to determine the I.H.P., B.H.P., and quantity of heat given up in the condenser. Indicator cards were taken from each end of the cylinder every fifteen minutes.

The method employed to determine these results was as follows: the engine was run with a load applied to the fly-wheel rim by means of a Prony brake, this load being kept constant and measured by the pressure of the brake upon a set of weighing scales; indicator cards were taken by means of a pair of "Thompson" indicators

connected to the ends of the engine cylinder; the revolutions of the engine were determined by means of a recording revolution counter, worked by a string attached to a pin in the end of the shaft; the quality and pressure of the steam in the supply and exhaust pipes were determined by means of two steam pressure gauges and calorimeters which took steam from the respective pipes; in order to determine the steam consumption of the engine, the exhaust steam was condensed in a surface condenser at atmospheric pressure and then weighed; the

quantity of heat given up in the condenser was determined by measuring the temperature of the condensed steam, the amount of condensing water, and its rise in temperature while passing through the condenser.

From the data thus obtained curves were drawn to show the following: quality of the steam in the cylinder at each point of the stroke during expansion; variation with time of the quantity of heat held by the steam in each end of the cylinder; variation with time of the amount of work done and also of the quantity of heat radiated to

the outside air; variation with time of the amount of heat supplied to the engine, and also the amount rejected by the engine. A study of these curves shows what probably happens in the cylinder to cause this initial condensation effect.

2- The brake used to apply the load was of the ordinary Prony type with four turns of rope around the fly-wheel, the frame being made of iron pipe. The zero reading of the weight of the brake on the scales was determined in the following

manner. The end of the brake being rested on the weighing scale, the engine is first turned, by hand, slowly forward and the reading of the pressure on the scales taken. The engine is then turned backward in the same manner and the reading of the scales again taken.

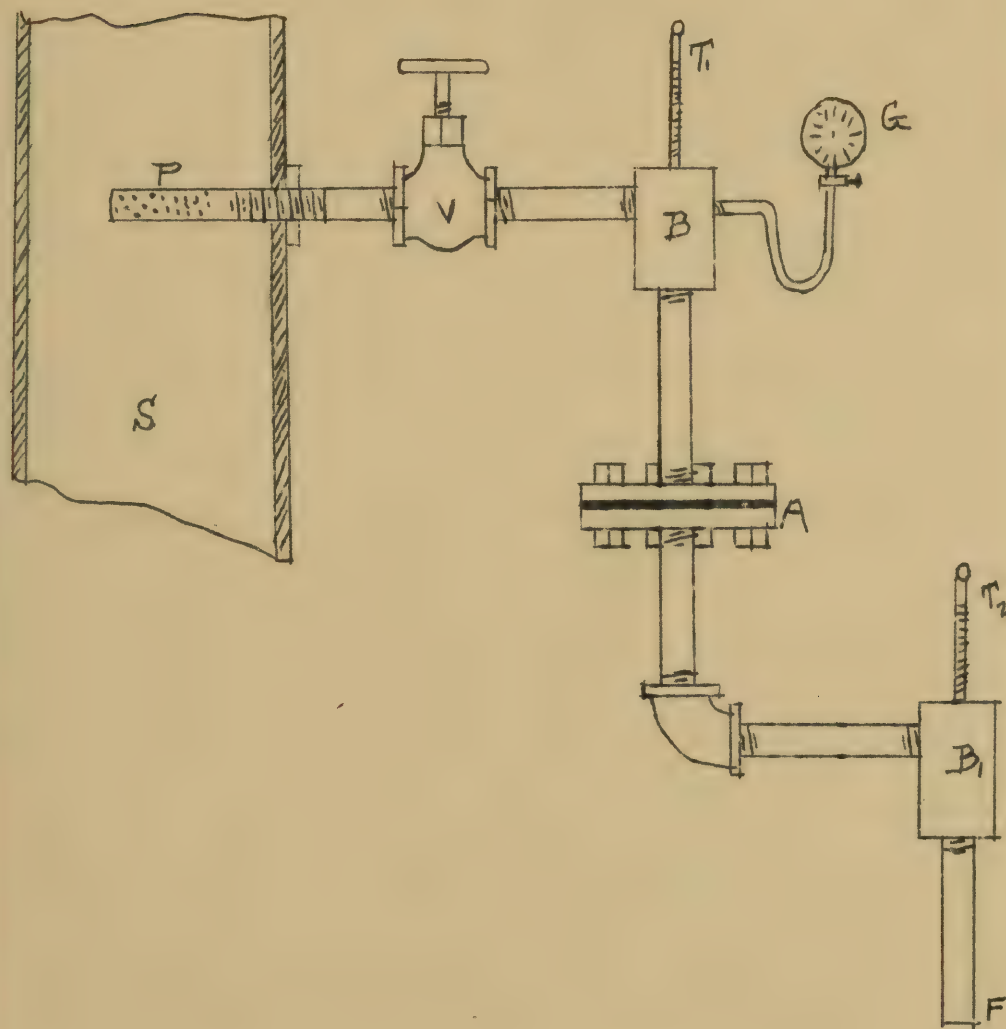
The mean of these two readings is the true scale reading when the engine is standing at rest. This method of getting the zero point of the scales, eliminates the effect of friction; since the friction force acts in the second case opposite to what it does in the first. Brake arm is 106 inches.

The revolution counter was

of the recording type, made by "Schaeffer and Budenberg" of New York. It was fastened to a wooden block that rested on the floor of the engine room. The counter was worked by means of a string which hooked over a pin in the end of the engine shaft. Readings of the counter were taken at stated intervals of time.

A "Barus Universal" steam calorimeter was used to get the quality of the steam supplied to the engine. This instrument is described in the "Transactions of A. S. M. E." vol. XI, p. 795. As the moisture in the steam used in this test was small, only the

heat gauge part of the calorimeter was used. a sketch and description of this is given below:



S is steam pipe which supplies engine.
 P is a sampling pipe which serves to take steam from pipe S. It is closed at the end, has many small holes in its sides, and extends into the centre of the pipe.
 B and B₁ are thermometer cups into which set two thermometers, T₁ and T₂.

V is a valve to control the steam supplied to the calorimeter.

G is a steam gauge which gives the pressure of the steam supplied. Let this pressure be equal to p_1 .

Steam to the calorimeter flows through an orifice at A and out into the atmosphere at F. Its pressure is reduced while passing through the orifice from that at G to the atmospheric.

The two parts of the calorimeter are insulated so that no heat is conducted by the pipe from the upper to the lower part of the instrument.

If, x = dryness of steam.

q = heat of liquid.

r = " " vaporization.

Then the heat in one pound of steam, above the orifice A, is:

$$q_{p_1} + x r_{p_1} = H.$$

After passing the orifice, the pressure being lowered, the steam is superheated by its own heat. Hence the heat in one pound of steam, below the orifice is:

$$q_a + r_a + .48(t_1 - t_2) = H.$$

Where a = atmospheric pressure.

$t_1 - t_2$ = degrees of superheat.

Equating these two heats, since it is assumed that no heat is lost in passing through the orifice, we have:

$$q_{p_1} + x R_{p_1} = q_a + R_a + .45 (t_1 - t_2) \quad (I)$$

t_2 = saturation temperature = 212°F .

t = temperature of steam at B .

The temperature read by the thermometer T_2 is however lower than t_1 , since the thermometer is not in direct contact with the steam. Similarly the reading of thermometer T_1 is lower than the temperature T of steam at the pressure p_1 .

It is assumed that

$$t_1 - t_2 = T - T_1$$

Therefore

$$t_1 = T - T_1 + T_2.$$

Hence from equation (I).

$$X = \frac{q_a + r_a + .48(T - T_1 + T_2 - t_2) - q_{fr.}}{r_{fr.}}$$

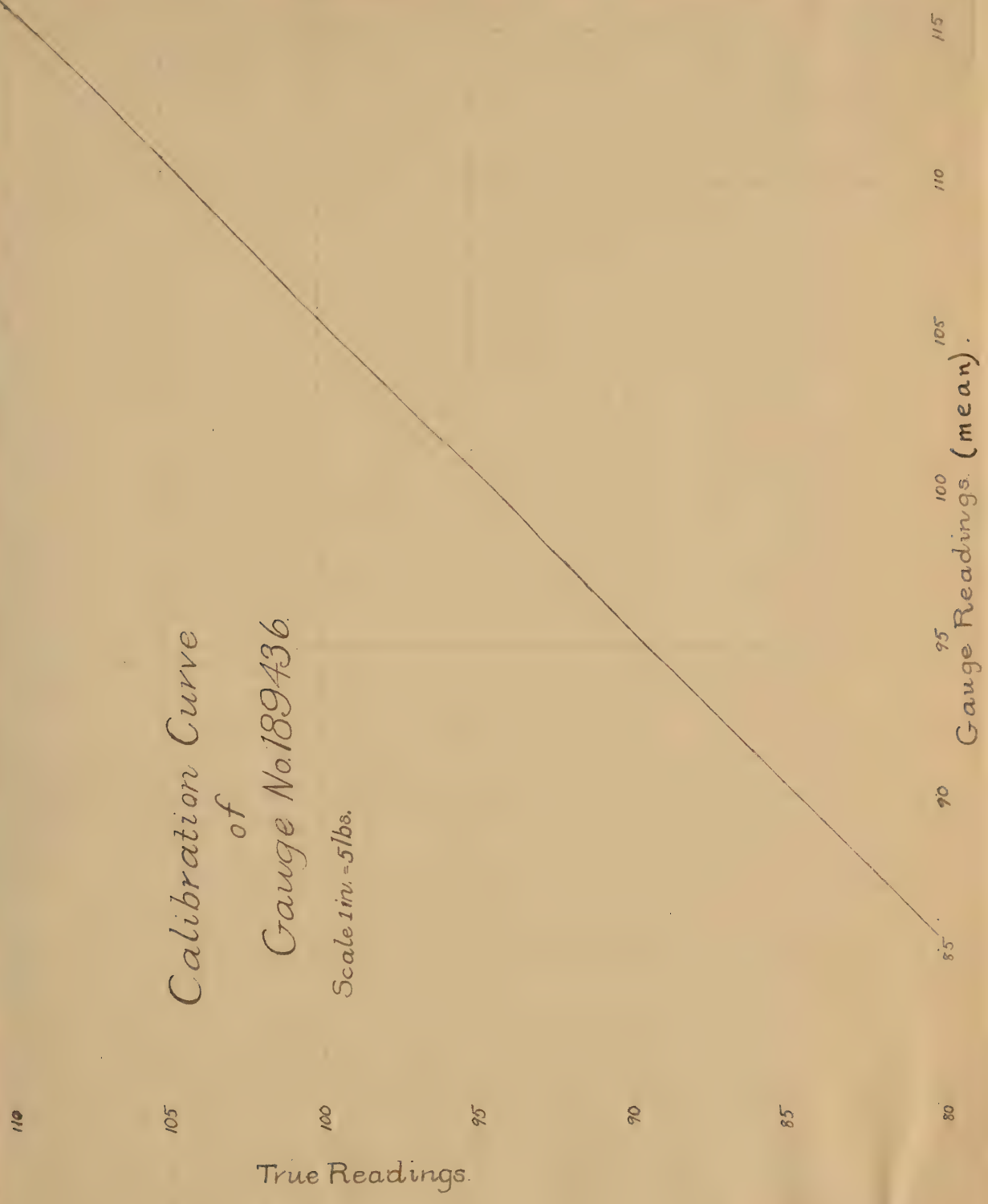
This gives the quality of the steam in the supply pipe.

The gauge, G , which was used to get the steam pressure p_1 , is "Burdon" steam gauge no. 187436, made by "Williamson and Carridy" of Philadelphia. Its scale is graduated in pounds from 0 up to 120 pounds, gauge.

This gauge was calibrated by means of a "Gorby" Steam gauge tester. From the calibration table the curve was drawn.

Calibration Curve
of
Gauge No. 189436.

Scale 1 in. = 5 lbs.



This gauge was calibrated for both ascending and descending readings, the mean being given on the curve as the "gauge-reading". The gauge pressures were corrected from this curve and the true pressures used in working up results.

A "Carpenter" separating calorimeter was used to get the value of x in the exhaust. It took steam from the exhaust pipe by means of a sampling tube similar to the one used with the Barnes calorimeter.

This instrument is completely described in "Carpenter's Experimental Engineering" 402, edition of 1897; also in the catalogue of "Schaeffer and Budenberg",

p. 105, edition of 1899. The
 water separated out is measured by
 means of a graduated scale placed
 alongside of a gauge glass con-
 nected with the inside of the
 instrument and in which the
 height of the collected water
 shows. Each scale division
 represents .01 pound of water.
 The amount of steam going
 through the instrument is
 measured by being passed
 through an orifice. The
 weight of steam flowing
 through is determined from
 the formula of Napier, which is:

$$W = \frac{h \cdot A}{70}$$

Where; h = absolute pres. of steam
 A = area of orifice.

When steam flows through an orifice of area A from a pressure P_1 to a lower pressure P_2 , the weight of steam flowing through in a given time is $W = P_1 \times K$, K being a variable.

The formula of Napier gives:
 $K = \frac{A}{70}$, when $W =$ pounds of steam through the orifice per second, and P_2 is less than $\frac{3}{5} P_1$.

As this calorimeter was used when P_2 was atmospheric pressure and P_1 was just a pound or two higher, it was necessary to calibrate this calorimeter to find how the value, K , varied with the pressure P_1 .

This was done by comparing the two gauge scales mentioned and described on page 19. This method gave the value of K for

values of P_1 above 25 pounds, absolute. For lower values of P_1 than this the following method of calibration was employed: the pressure P_1 was measured by means of a mercury manometer; the steam discharged through the orifice, A, was condensed and weighed.

Then by taking the time of discharge it is very simple to find the mean rate of discharge through the orifice A for various low pressures, P_1 .

The curve on page 18-C shows the variation of K with the pressure P_1 .

In using the values from this curve, W = pounds of steam discharged in 10 minutes.



The gauge attached to this calorimeter had two scales, one reading pounds pressure, and the other reading pounds of steam through the orifice in ten minutes.

From a comparison of the two scales, the constant $\frac{A}{70}$, was found to be .04; where W is the number of pounds of steam through the orifice in ten minutes. The gauge was graduated down to about 20 pounds absolute. As the pressure in the exhaust was only about 15 pounds (absolute), a mercury manometer, graduated in inches, was used to measure the steam pressure at the calorimeter. At the lowest graduation on the

gauge scale, the constant was found to have dropped only to .038 instead of .04.

Because of this fact, and since it was found that a very considerable error in the amount of steam through the orifice made but slight error in the value of x , it was assumed that the constant, $\frac{A}{70}$, was always equal to .04, even at the low pressure of 15 pounds absolute.

If, n = pressure in inches of mercury,
then $\frac{n}{2}$ = pressure in pounds.

14.7 = atmospheric pressure.

$\therefore \left[14.7 + \frac{n}{2} \right] .04 = W$, the pounds of steam through the orifice

in ten minutes.

m = weight, in pounds, of the water separated out by the calorimeter in ten minutes, H is determined from readings of the height of the water in the gauge glass.

Then we have:

$$x = \frac{W - m}{W + m} = \text{dryness of steam.}$$

The condensed steam from the condenser was caught in a tank placed upon a set of weighing scales. By reading these scales at stated intervals of time, the weight of steam taken by the engine per minute is easily determined.

The condensing water was measured by being passed over

a standard weir having end contractions. The head of water on the weir was measured by means of a hook gauge placed in the weir flume. The weir used was 10 inches wide. Zero reading of the hook gauge was taken when the water level in the flume was just at the crest of the weir.

Formulas used in connection with the weir were those of "Francis":

$$Q = 3.33 (l - .2h) h^{\frac{3}{2}}$$

Q = cu. ft. of water over weir per second.

l = width of weir, in feet.

h = head of water on weir, in feet.

Thermometer cups were placed in the pipes just

outside of the condenser. By means of thermometers placed in these, the following temperatures were taken; condensed steam, condensing water entering condenser, condensing water leaving condenser.

These thermometers were graduated to read Fahrenheit degrees. From these temperatures, knowing the amount of condensed steam from the condenser and also the amount of condensing water, it is a very simple matter to find the amount of heat given up or accounted for in the condenser.

The indicators used to take cards with were two "Thompson Improved" instruments,

made by the "American Steam Gauge Co." of 43 Astor. This form of indicator is familiar to every engineer, so a description of it is considered unnecessary. However, a very good description is given in the catalogue of "Schaeffer and Budenberg", page 78, edition of 1899. Only a few details are different in this description from those on the actual instruments used in making this test. These indicators were fastened on to cocks set directly in the sides of the engine cylinder, with no pipe connections whatever.

The springs used in these indicators were 60 pound "Thompson"

springs, numbered the same as the indicators themselves.

Indicator & spring no. 3257 was used on the head end of the engine cylinder, and no. 3071 on the crank end.

These springs were calibrated in their respective indicators by a method of comparison with a steam pressure gauge. This method is described in "Carpenter's Experimental Engineering", page 479, edition of 1897. It consists in applying the same ~~steam~~ pressure to the indicator cylinder and to the steam gauge, thus comparing the gauge reading with the amount of movement of the indicator pencil. The steam pressure was

made to gradually rise, starting at the atmosphere after the instruments had become heated up. For every increase of 10 pounds in the steam pressure as read by the gauge, the indicator drum was turned, thus making the pencil draw a horizontal line. This gave the "ascending" calibration.

A similar process was employed with the pressure gradually descending again to the atmosphere. These horizontal lines show the amount of vertical movement of the indicator pencil corresponding to the different steam pressures as read by the gauge.

These horizontal line calibrations are given on pages 29 and 30, the strain pressures marked being those read by the gauge used, which was of the "Bourdon" type, no. 159434, made by "Williamson and Cassidy" of Philadelphia. This gauge was then tested on a "Crosby Strain Gauge Tester" and its ascending and descending calibration curves drawn as on page 31. By measuring the amount of spring compression (actually, pencil movement) from the diagrams on pages 29 and 30, and then correcting the strain pressures by means of the curve on page 31, we are enabled to

draw both "ascending" and "descending" curves showing the variation of spring compression (pencil movement) with the true steam pressure under the indicator piston. These curves are then used to correct the indicator cards taken by a method to be described later.

These calibration curves are given on pages 32 & 33 of this thesis.

Calibration of Indicator Spring No. 3257

Pres. (lbs)	Compression (In)
95	1.44
90	1.35
80	1.18
70	1.02
60	.84
50	.66
40	.49
30	.35
20	.22
Atm.	0

Ascending Readings.

Pres.	Comp.
95	1.50
90	1.42
80	1.27
70	1.10
60	.93
50	.75
40	.59
30	.41
20	.24
10	.06
Atm.	0

Descending Readings.

Calibration of Indicator Spring No. 3071.

Pres. (lbs.)	Compression (in.)
89	1.51
80	1.34
70	1.17
60	.99
50	.80
40	.64
30	.45
20	.27
10	.09
Atm.	0

Ascending Readings.

Pres.	Comp.
85	1.44
80	1.35
70	1.19
60	1.00
50	.82
40	.64
30	.44
20	.22
10	.06
Atm.	0

Descending Readings.

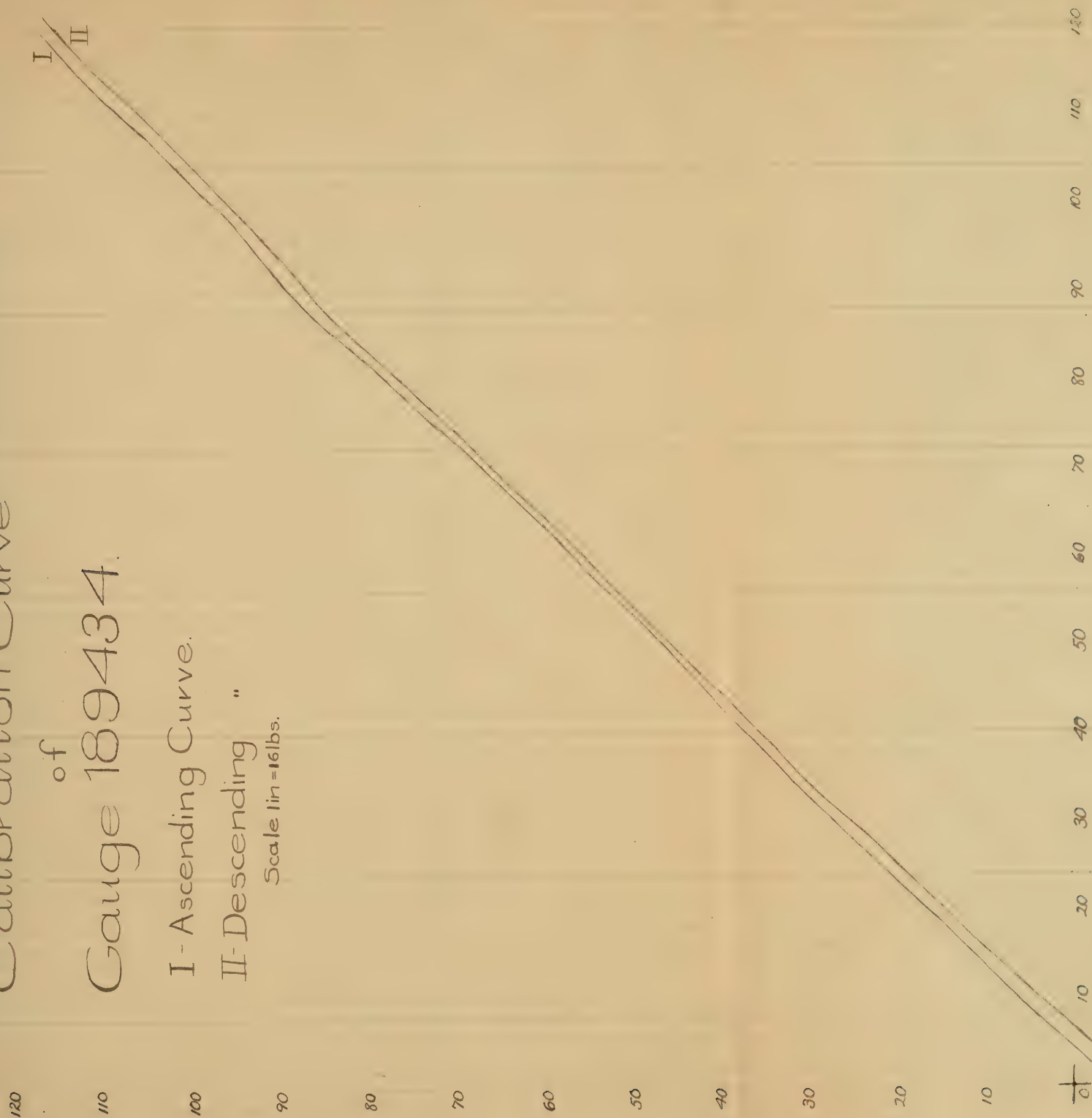


Calibration Curve of Gauge 189434.

I - Ascending Curve.

II - Descending "

Scale 1 in = 16 lbs.



True Readings.

Gauge Readings.

Calibration Curves of

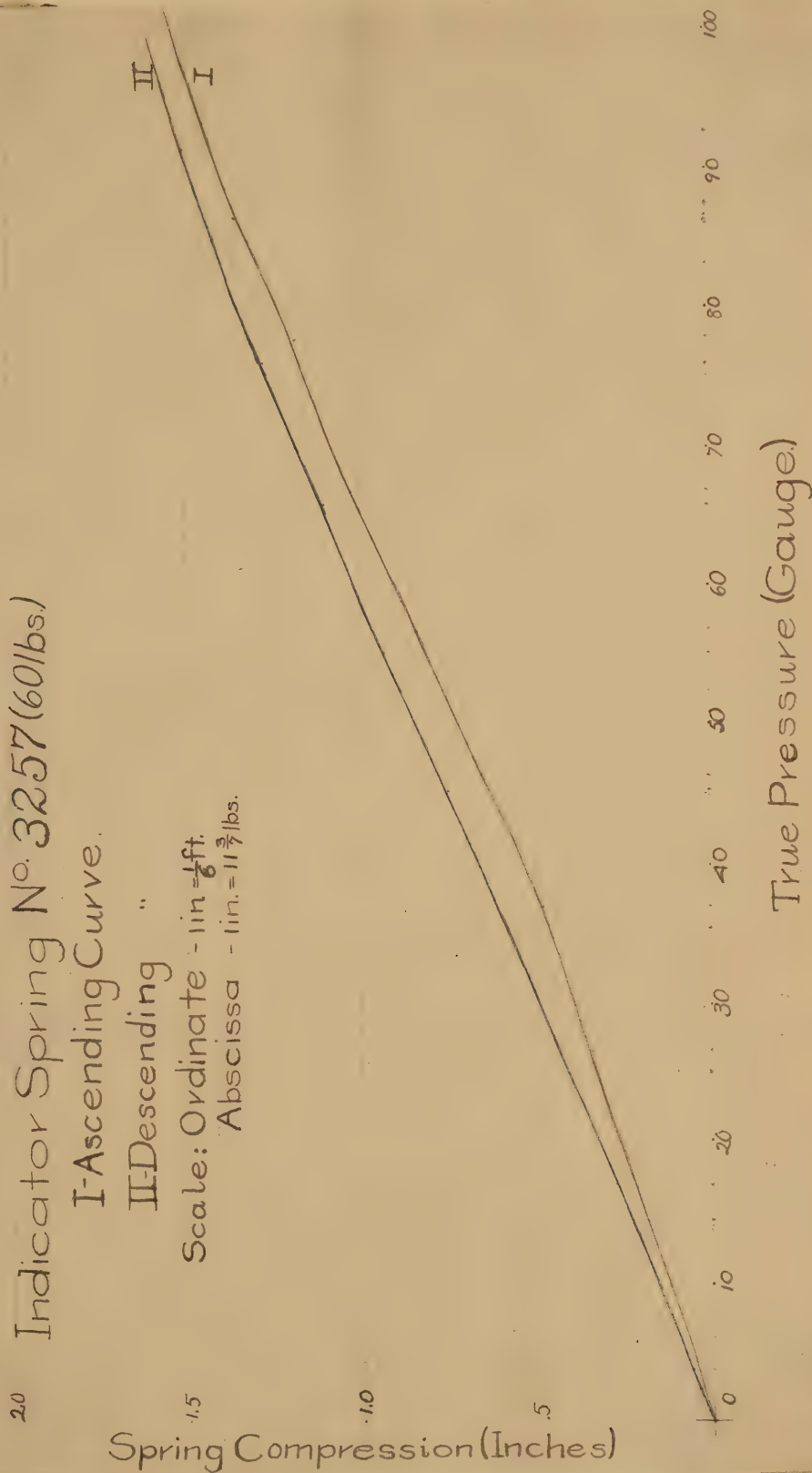
Indicator Spring No. 3257 (60 lbs.)

I-Ascending Curve.

II-Descending "

Scale: Ordinate - $1 \text{ in} = \frac{1}{8} \text{ ft.}$

Abscissa - $1 \text{ in.} = 11 \frac{3}{4} \text{ lbs.}$



Calibration Curves

of
Indicator Spring No 3071 (60 lbs)

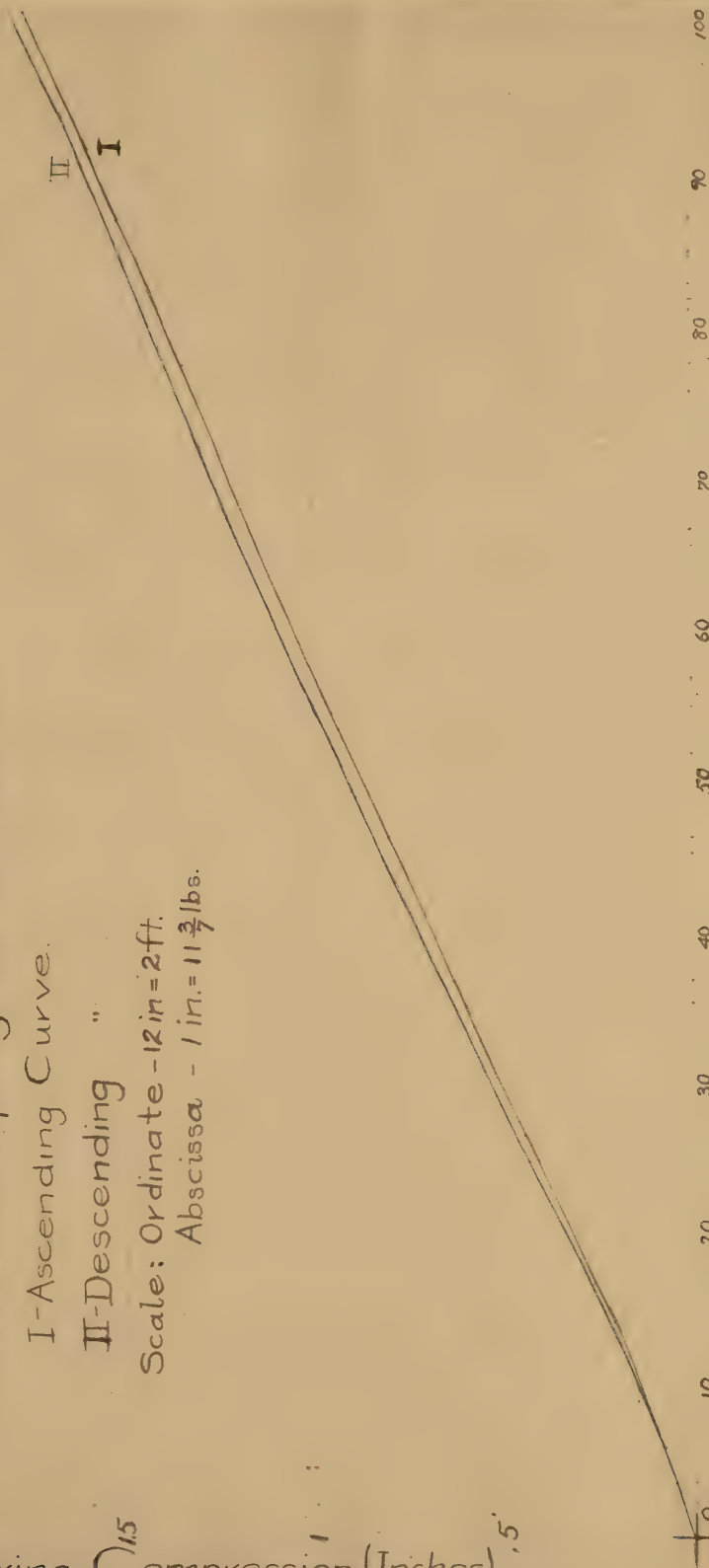
I-Ascending Curve.

II-Descending "

Scale: Ordinate - 12 in = 2 ft.

Abscissa - 1 in. = $11\frac{2}{3}$ lbs.

Spring Compression (Inches)



True Pressure (Gauge)

3-The engine used to make these tests is a horizontal, single expansion machine of the "Porter-Allen" type, made by the "Southwark Foundry and Machine Co." of Philadelphia.

It is rated at 45 horse power with $\frac{1}{4}$ cut off and 280 rev. per minute. The speed has, however, been geared down to about one half of that at which it is rated. For a complete description of this engine and its parts, I would refer to the thesis by "Witness" and "Reeder" of '1900' class.

The clearance volume of this engine was found by placing the engine on its dead centre and then pouring in water until this

clearance volume was filled, noting the time taken to pour and the amount of water poured. Of course some of this water leaks out and thus must be corrected for. This is done in the following manner: the time at which the clearance volume is full of water is noted; the water is then allowed to leak for a short while; then more water is poured in until the clearance volume is again full, the time being again noted. By knowing the amount of this extra water poured in, we thus know how much has leaked out between the times at which the clearance volume is full, and from this can get the rate of leakage which is used to correct our first-

result. Then when we have found the corrected quantity of water taken to fill the clearance volume it is very simple to find this volume in cubic feet.

From several tests made in the above manner the following mean results were obtained.

Head end clearance = .0308 cu. ft.

Crank " " = .0316 " "

Below are given the other constants of the engine.

Inside cylinder diameter = $8 \frac{1}{2}$ "

Piston rod " = $1 \frac{1}{2}$ "

Area head end of piston = $50.64 \square$ "

" Crank " " = $48.88 \square$ "

Length of stroke = 16"

Length of connecting rod = 48"



4- I am working up the results of this thesis, Peabody's tables of "Saturated Steam" were very extensively used. The following is the notation used in these tables and in this thesis.

p = pressure, pounds per square inch, absolute.

t = temperature, degrees, Fahrenheit.

q = heat of liquid, B.T.U.

λ = total heat in " "

h = heat of vaporization.

ρ = " " internal work.

$A p u$ = " " external " "

$\int \frac{cdt}{T}$ = entropy of liquid.

s = specific volume of steam.

r = weight in pounds of 1 cu. ft. of steam.



It would also probably be well to refer to "Spangh's Notes on Thermodynamics" for many of the principles of heat which are used here.

5- The following test results were calculated in the manner given below.

Revolutions per minute (R.P.M.) is the number of revolutions recorded in a given time, divided by that given time in minutes.

Quality of boiler steam (x_1) is found by the formula on page 15.

Quality of exhaust steam (x_2) by formula on page 21.

Steam rate per minute is found by dividing the weight of condensed steam discharged from the condenser in a given time, by that given time in minutes.

Steam rate per revolution is the steam rate per minute divided by the rev. per minute.

The amount of steam supplied to the engine per revolution is assumed to be divided between the two ends of the cylinder proportionally to the areas of the corrected indicator cards.

Pressure of supplied and exhaust steams is found by correcting the pressures read by the respective calorimeter gauges.

Mean Effective Pressure (M.E.P.)

the cards being corrected for spring pressure, is found by the following relation or formula:

$$M.E.P. = \frac{\text{area of card, sq. in.}}{\text{Length of card, ins.}} \times \text{Spring scale.}$$

Indicated horse power (I.H.P.) is found thus:

$$I.H.P. = \frac{P L (A_h + A_c) N}{33000}, \text{ where}$$

P = mean eff. pressure, M.E.P.

L = Stroke of engine, in feet.

A_h = area head end of piston, sq. ins.

A_c = " crank " " " " "

N = Rev. per minute = R.P.M.

Brake horse power (B.H.P.) is the power delivered by the engine and measured by the Prony brake. It is

calculated as follows:

$$B.H.P. = \frac{l \cdot h \cdot 2\pi N}{33000}$$

l = brake arm, in feet, = 106".

h = pressure, in pounds, on
brake scales = brake scales

reading less their zero reading.
Zero reading with brake
resting on scales, and determined
by the method on page 9, is
equal to 66 pounds.

N = Rev. per minute as
found above.

Mechanical efficiency
is found thus:

$$\text{Mech. Eff.} = \frac{B.H.P.}{I.H.P.}$$

It is necessary for
this thesis to know the
amount of strain in the

cylinder during expansion. This is equal to the amount supplied per cycle added to that which is in the cylinder during compression.

The amount of steam in the cylinder during compression is determined in the following manner: from the corrected indicator card, we measure the pressure and volume of the steam in the engine cylinder at the beginning of compression; next, assume that at this point the quality of the steam in the cylinder is the same as that in the exhaust. We then know the quality, volume, and pressure of this steam which is in the cylinder during

compression. By using Peabody's "Steam Tables" we can easily find the weight of steam in the cylinder during compression, as follows:

$$m = \frac{V}{x \cdot S_p}$$

S_p = specific volume at pressure measured

V = volume measured.

x = dryness of steam.

By means of the formula herein given, the following table of "test results" has been calculated. The first table gives the results ordinarily calculated for an engine test.

The second table gives results and data needed to work this thesis.

Table I

	Test I	Test II
Rev. per minute	168	165
Steam supplied per minute, pounds.	28.8	21.2
Mean. Eff. Press.	40.3	32.5
Ind. horse power.	27.15	21.55
Brake " "	22.3	16.3
Mechanical Efficiency.	82.1%	75.9%
Pounds of steam per I.H.P. hour, dry.	63.3	58.8
Pounds of steam per B.H.P. hour, dry.	77.1	77.7
Pounds of condensing water per minute.	312.6	312.6

Table II.

	Test I	Test II
Absolute boiler pressure.	118.8	115.0
Absolute exhaust pressure.	16.01	16.11
Head end back press. (From card)	20.5	18.2
Crank end back press. " "	20.5	18.2
Value of X in supply pipe.	.995	.996
" " " " exhaust	.974	.953
Area corrected card. (Head)	4.780"	3.940"
" " " (Crank)	4.890"	3.850"
Steam per revolution (lbs)	.1713	.1285
Steam to head end, supplied (lbs)	.0847	.0649
" " crank " " "	.0866	.0636
Steam in head end during compression	.00612	.00627
" " crank " " "	.00662	.00634
Total steam in Head cycle.	.0908	.0712
" " " Crank " "	.0932	.0699

6. Diagrams and explanations (see pages following.)

A, B, E, & F are simple tracings of the actual indicator cards which were taken.

A = "head" end card (mean) from test I.

B = "crank" " " " " " I.

E = "head" " " " " " II.

F = "crank" " " " " " II.

The atmospheric pressure line is shown on all of the diagrams.

C, D, G, & H are these same diagrams or cards, A, B, E, & F, with their corners squared. This is done by simply drawing in, even smooth lines and curves in place of the somewhat irregular ones of the actual cards.

By means of these square cards, we can show the important points of the stroke distinctly. Having these diagrams, C, D, G, & H, we then draw a series of horizontal lines at the proper height above the "atmosphere" so as to correspond to and represent the pressures as taken from the spring calibration curves on pages 32 and 33. These lines are shown in red, the "ascending" spring curves being used to place them on the compression ends of the cards, and the "descending" spring curves for the expansion ends. This is done thus because the indicator piston and pencil are moving upwards while making the compression ends of a card and downwards while drawing

the expansion curve.

The corrected gauge pressures are marked on the diagrams opposite the proper pressure line.

Calibration curve of spring no. 3259 is always used for cards taken from the head end of the engine cylinder, and curve of spring no. 3091 for cards from the crank end.

A, B, C, & D belong to test I

E, F, G, & H " " " II

The length of these cards as measured is:

$$\text{mean length} = 4.2''$$

Volume scale of these cards is thus:

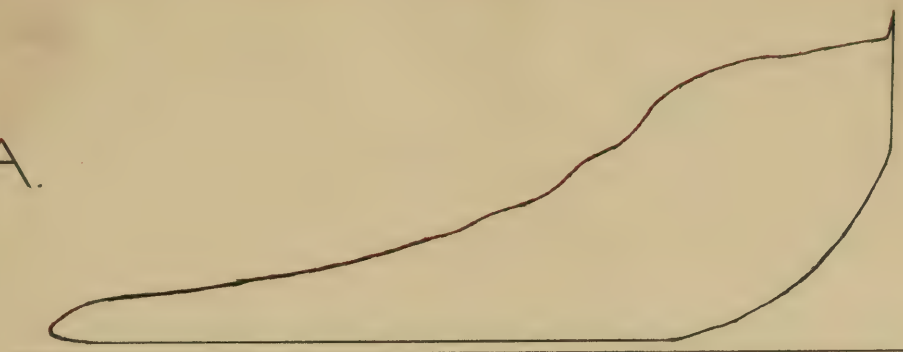
for head end cards,

$$1'' = \frac{16}{4.2} \times \frac{50.64}{1728} = .01115 \text{ cu. ft.}$$

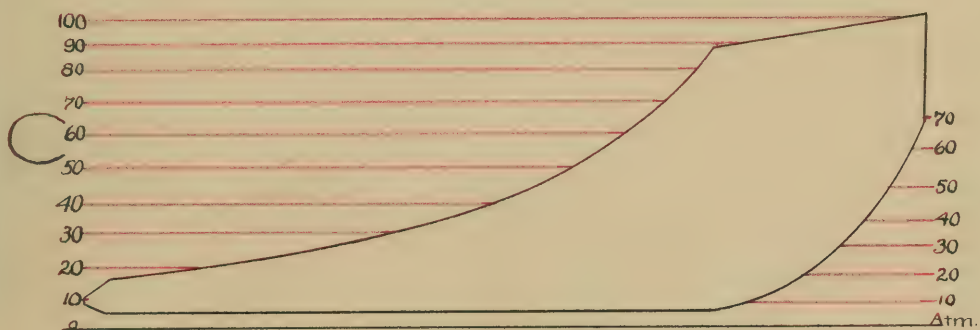
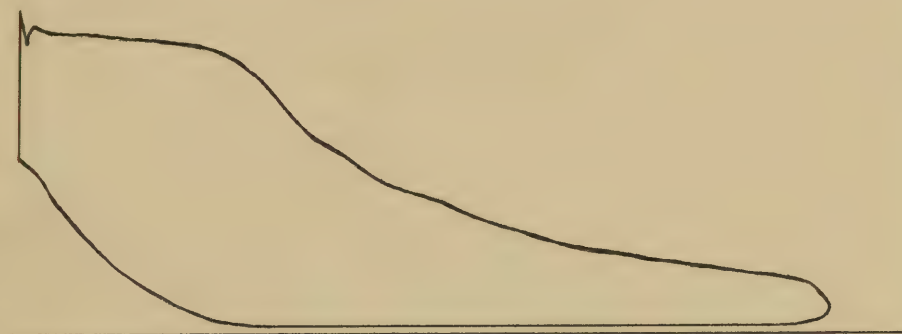
for crank end cards.

$$1'' = \frac{16}{4.2} \times \frac{48.88}{1728} = .01085 \text{ cu. ft.}$$

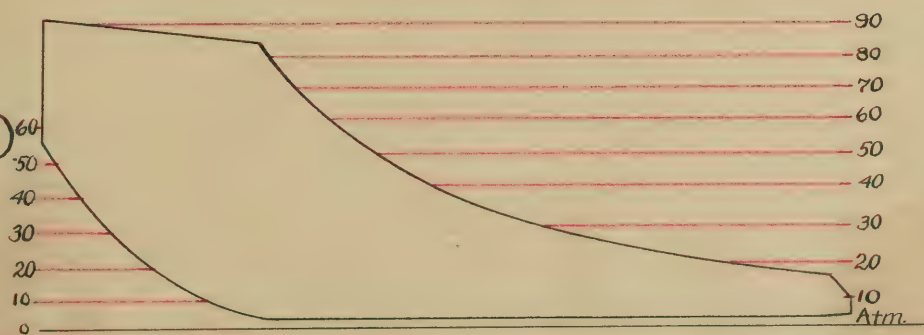
A.



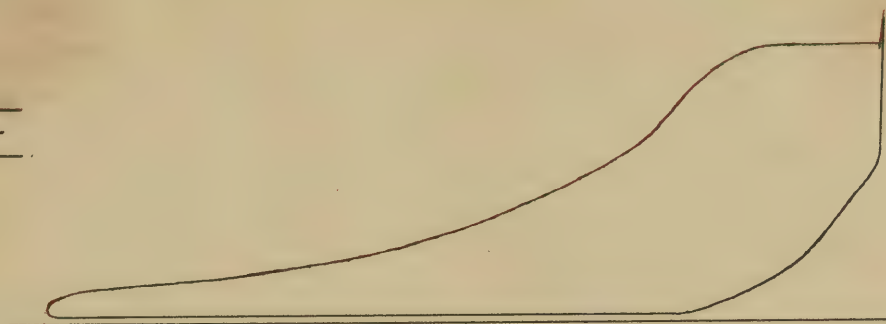
B.



D



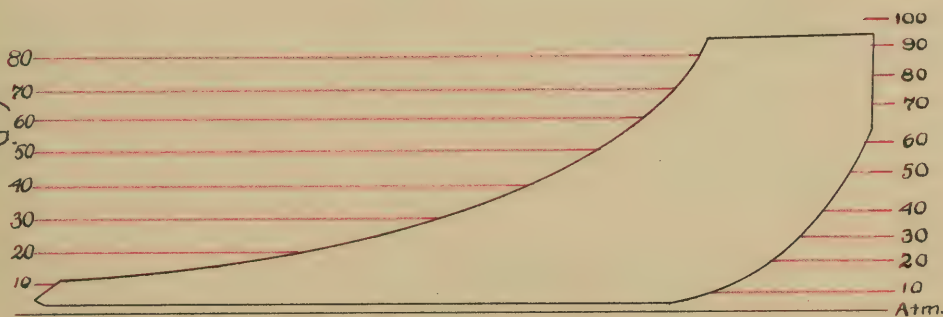
E



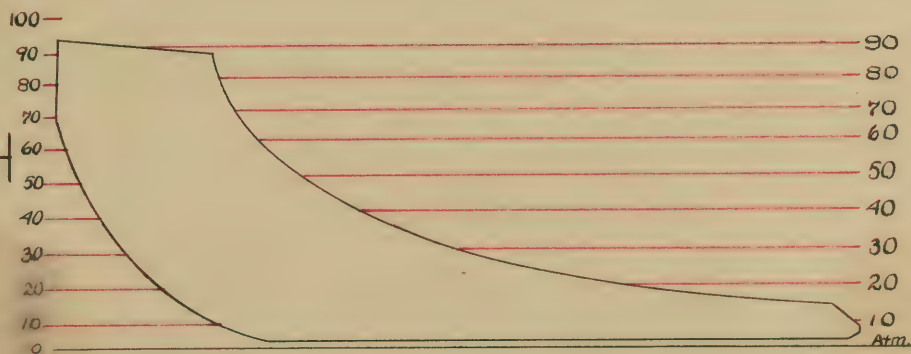
F



G



H



Having now these square cornered cards, C, D, G, H, we proceed to correct for the variation in the indicator spring compression. At the same time that this is done, however, for convenience we changed the scale of the cards so as to have a uniform pressure scale of 50 pounds to 1 inch compression and a card length of 6 inches. For these corrected cards, see diagrams K, L, N, R on pages 58 to 61.

The method of constructing these corrected cards is as follows (see diagram K, page 58 for reference). Diagram K is the corrected drawing of card C, page 49. Referring now only to the upper part of K; lay off a horizontal distance $a-b$ equal to 6 inches,

this representing the cord length or stroke to a scale of:

$$1" = \frac{16}{6} \text{ inches.}$$

At the ends of the stroke ab draw the two vertical lines ac and bf.

Starting now with the line ab as the absolute zero of pressure, lay off on ac the distance ad equal to the atmospheric pressure, 14.7 pounds, to a scale of $1" = 50 \text{ pounds}$.

Through d draw the horizontal line shown dotted red and marked "Atm." Then using the vertical scale of $1" = 50 \text{ lbs.}$, draw the series of horizontal lines, shown dotted red, to represent each increase of 10 lbs. in the true gauge pressure. We are then ready to construct the diagram K from the diagram C.

The horizontal or volume scale

of the diagrams K, L, N, R is then:

$$1'' = \frac{16}{6} \times \frac{50.64}{1728} = .0780 \text{ cu. ft.}$$

That is, cards from both ends of the engine are reduced to the scale of a head end card, the head end piston area being 50.64 in^2 .

The volume occupied by the steam at the different pressures on the card C is measured along the horizontal volume lines, to the scale of the diagrams C, D, G, H. Then these volumes are laid down, to the new volume scale, on their respective pressure lines of the diagrams K, L, N, R. Then through the points thus obtained, we draw the corrected indicator cards as shown.

Having now the corrected-

indicator cards with the absolute zero pressure line drawn below, we make a series of horizontal lines, shown full red, to represent each increase of 10 lbs. in the absolute pressure.

Then lay off the horizontal distance ao equal to the clearance volume, to the new volume scale, $1" = .078 \text{ cu. ft.}$ Draw the vertical line oe, this representing the line of zero volume.

Now we have from the table on page 45, the total weight of steam, M , in the cylinder during expansion. Then by means of the Steam Tables, we can find for each absolute pressure, the volume which would be occupied by this

quantity, M , if it were all dry steam, i. e. $x = 1$. Thus we are able to get points through which to draw the curve l_n , which is the expansion curve of dry steam. Then having the expansion curve l_n , of M pounds dry steam, and also the full black line r_s , which is the actual expansion curve of M pounds of steam; then at any pressure P , the ratio of the volume hP to the volume hV is the value of x at the point P , i. e. the quality of the steam at that point of the stroke.

Below each of these corrected cards in the diagrams K, L, N, R , is drawn a curve showing the

the variation of this quantity x during the expansion part of the stroke. The abscissae of this curve show points in the stroke and the ordinates, the value of x . The value of the ordinates of this curve are shown by the horizontal red lines drawn with the corresponding value of x marked thereon. Scale is: 1" ordinate = .1 in value of x .

Thus knowing as we do, the weight of steam M in each end of the cylinder during expansion and also the pressure and quality of this steam at any point of the stroke during expansion, we are able by means of the Steam Tables to calculate the quantity of heat,

above $32^{\circ}7$, actually in the stream of each end of the cylinder during any part of the stroke between cut-off and release.

Diagrams are as follows:

K = C corrected, with additions as above.

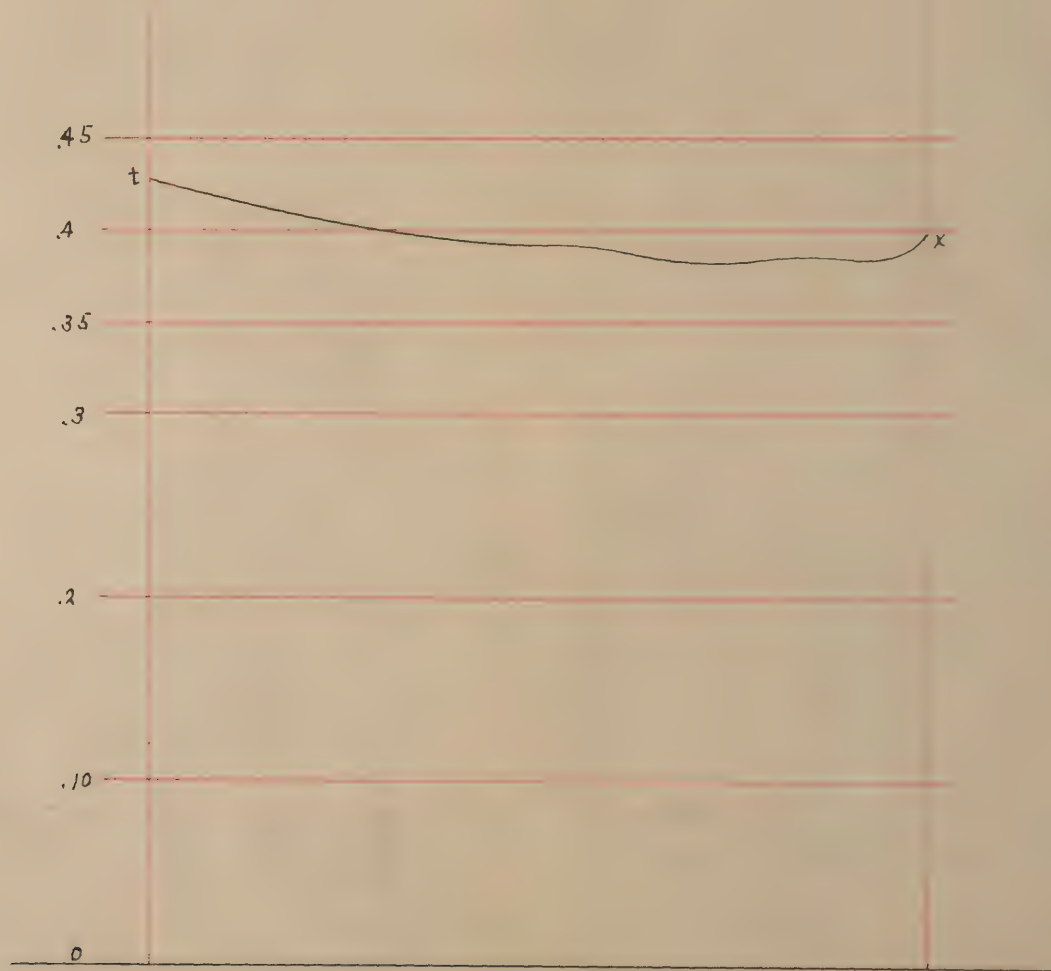
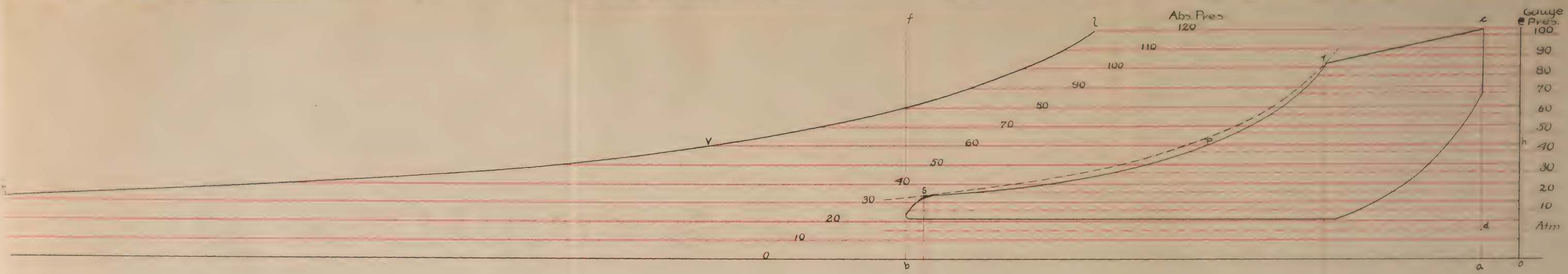
L = D " " " "

N = G " " " "

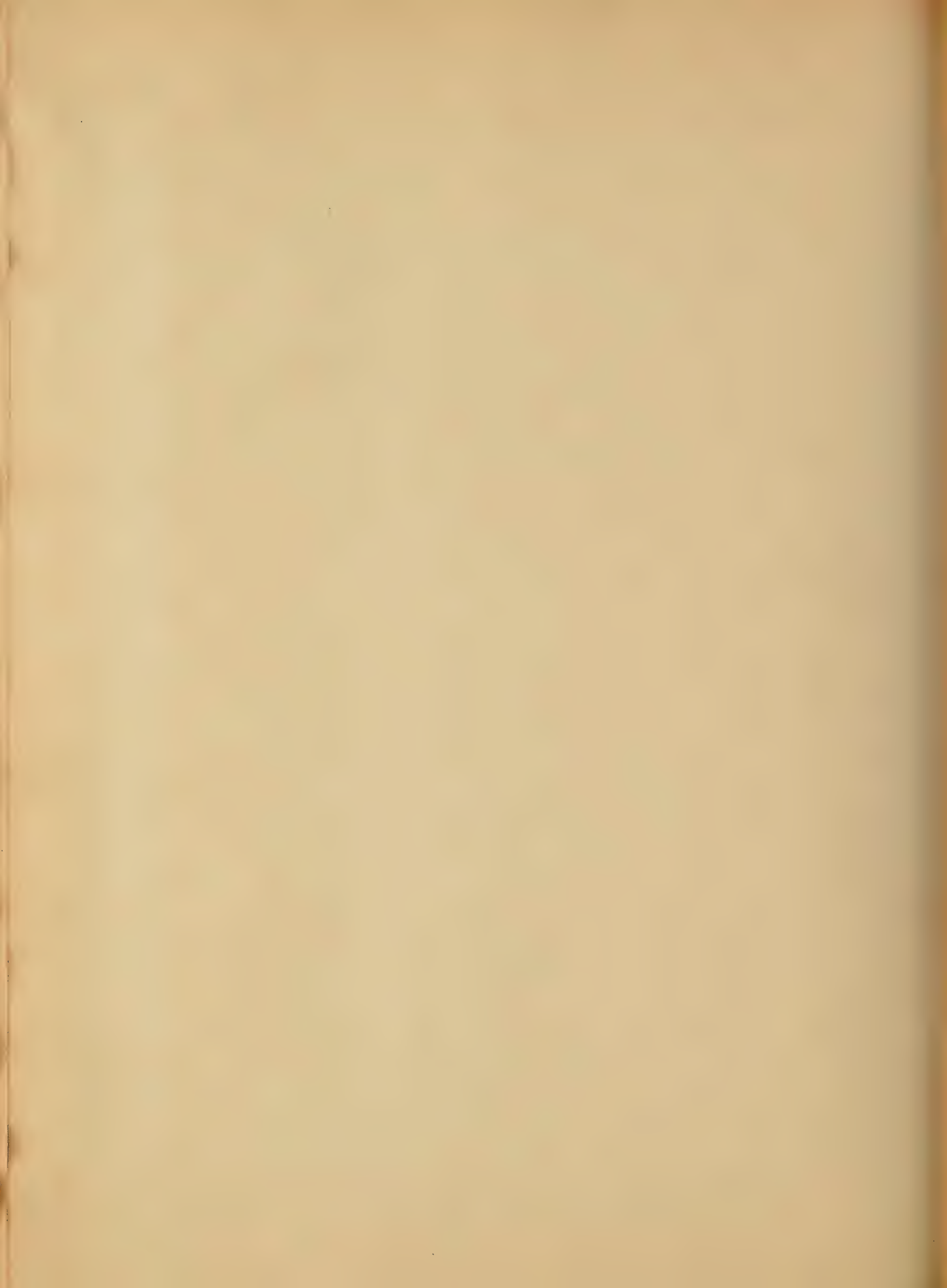
R = H " " " "

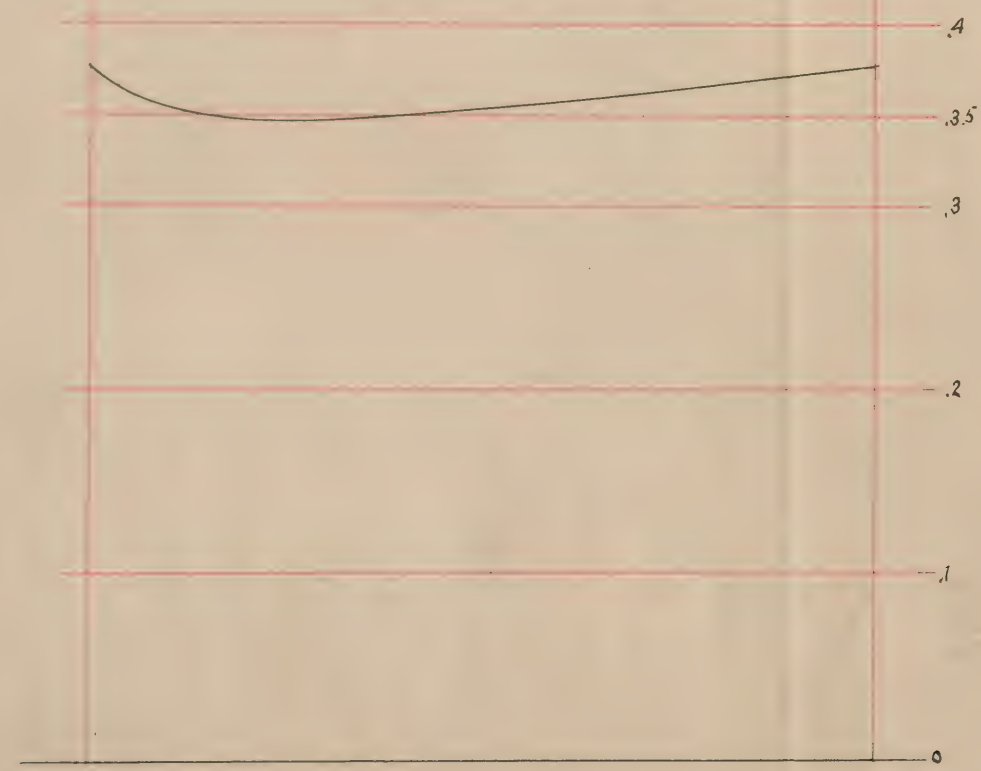
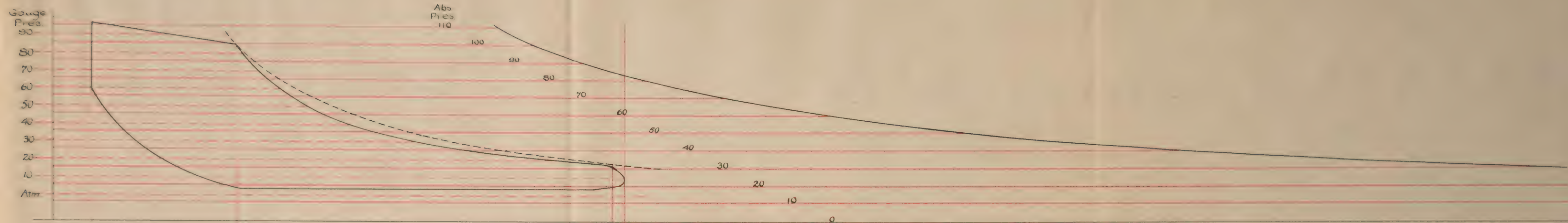
The dotted black curve RS is the rectangular hyperbola drawn through point R.

The rise in the x curve of diagram R, towards the end of the stroke, is probably incorrect, it being due to a leaky steam valve. This is probable since the expansion curve rises so much above the rectangular hyperbola.

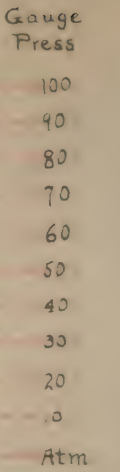


K.





L.



45 -

.4

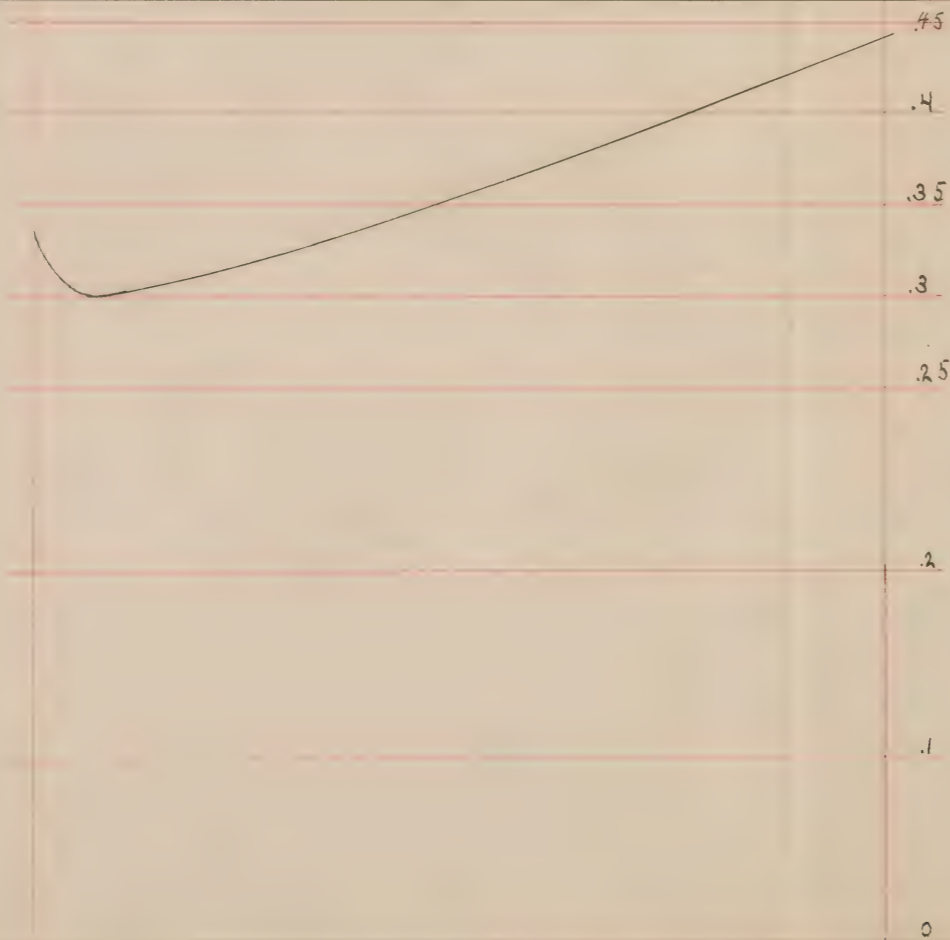
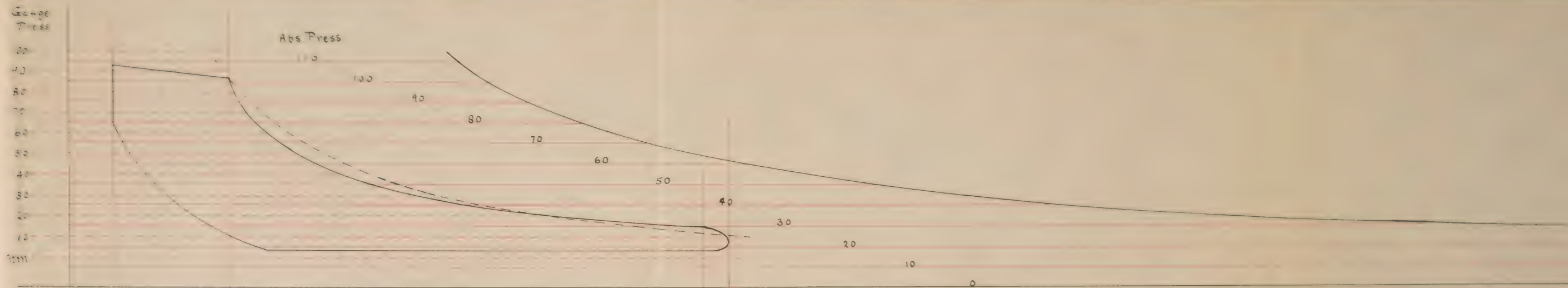
.35

.3

.2

1

0



R.

7- As we wish to show the rate of heat changes in the steam engine, it is necessary to draw curves between heat and time.

In the curves here drawn, the abscissae represent time and the ordinates, British Thermal Units. In order to change from points along the stroke to a time axis, we first assume that the engine turns at a uniform rate during the whole revolution.

In drawing these time heat curves, only the first test (I) is "worked up" and its curves constructed. Test II was used to help construct the time-heat curves for test I.

The engine, in this test, ran at the mean rate of



168 R.P.M., with a cut-off of about $\frac{1}{4}$. There are 360° in one revolution, therefore the number of degrees through which the crank turns in one second is

$$\frac{168 \times 360}{60} = 1008.0 \text{ degrees.}$$

In .01 second, then the engine crank turns through $10^\circ 4'$.

Referring now to the diagram T on page 66, suppose we have the crank circle drawn to a given scale. Here the scale is $1" = \frac{16}{6}$ inches, and the crank circle is separated into two parts in order to give clearance to the diagram. Knowing that the crank turns through $10^\circ 4'$ in .01 second of time, lay off points on the crank circle

corresponding to every .01 second, starting with the engine on its "head" centre at a zero time.

From the point at which the crank is at any given time, with a radius equal to the connecting rod length, draw an arc cutting the stroke line. The point thus found on the stroke line is the piston position corresponding to the given ~~given~~ time with which we started. By this method points are found on the stroke, corresponding to every .01 second. On the other hand, starting with any point in the stroke we can work backwards and get the point on the crank circle and also the time corresponding.

We then lay off a horizontal time axis on which $1'' = .02$ seconds, calling the point of head end admission (i. e. the "head" dead centre) the point of zero time, i. e. the origin from which we start.

An inspection of diagram T will make clear this method as above explained.

T





8- Diagrams W and Y represent the variation of heat with time, page 79. Both have the same scale and zero point of time but the zero points for heat are not the same in both even though the heat scale is.

Diagram Y shows the variation of heat held by the steam in the cylinder. Curve a b c d e a is for the "head" end steam and is shown in black ink. Curve f g h i k f is for the "crank" end and is shown in blue. These curves are both gotten in the same manner and by the same method so only one explanation is necessary.

First a horizontal time axis is laid off, the scale here being $1" = .02$ seconds. We suppose that the point of "head" admission, i. e. head

dead centre is the zero or starting point of time. Then from the stroke diagrams, as on pages 58 and 59, by means of the method explained in paragraph 7, we can put down on this time axis all of the important points of the stroke, cut-off, release, compression, etc. These are shown by the "dotted red" lines and are marked on the diagram.

We know from table II, page 45, the weight of steam in each end of the cylinder during expansion, and also from diagrams K and L, the pressure and quality of this steam at any point of the stroke between cut-off and release. Then by means of the Steam Tables and the simple formula of heat which



is: $H = m(q_f + x_f p_f)$, we have the amount of heat, above 32°F , actually in this steam at any point of the stroke between the cut-off and release. Hence we can construct to any suitable scale the curve bc showing the variation of heat in the steam during expansion. The heat scale here used is $1" = 10 \text{ B.T.U.}$, the zero line or being the heat at 32°F .

Knowing the weight of steam in the cylinder during compression, as in table II, we suppose that right at the point of compression this steam is of a quality equal to that of the exhaust and of a pressure equal to the "back

pressure" measured from the card. We can then calculate the heat held by the steam at this point, thus getting the point c. By measuring from the card the work of compression we get the heat added during compression and thus get the curve from c to a. This of course assumes that the compression is adiabatic.

Now we see that the curve rises from a to b and drops off from c to e. Let us make a test (II) having admission at σ and cut-off at u'. We can then get weight of steam w which has come in the cylinder from σ to u', and also its quality at cut-off, w'.

Now we assume that in test I,

the weight and quality of the steam in the cylinder at the time m' are the same as at this time (n') in test III. Thus by making a series of tests II, III, IV, having cut-off points earlier than in test I, we are able to get the points m, n, s and are then able to draw the curve a b.

Very little can be fairly assumed or found out about the curve from c to e. If however the steam expands freely from the point of release to the end of the stroke, knowing the volume and pressure at the end of the stroke, we can calculate for a point d. This assumption of "free expansion" is really a fair one to make for the

following reasons: we see that during the latter part of expansion, that is along the curve b-c, the heat held by the steam is nearly constant, i. e. the cylinder walls are not suddenly giving up any large amount of heat; can we not then suppose that this amount of heat held by the steam would have still remained nearly constant if release had been right at the end of the stroke. Therefore, knowing the quality and pressure at release and also the pressure at the end of the end of the stroke, we can get the quality at this last point, assuming, as it were, "free expansion" and neglecting the work, which

is very small. The curve from d to e is then a mere assumption or guess. It is simply a continuation of c-d which is curved around so as to meet e-a.

We now wish to explain the upper diagram, W, on which there are five curves. Here let the horizontal line OP represent the amount of heat, above some zero point, which is in the whole engine at the point of zero time, i. e. the lead and admission. Curve o, x, t, u, v, q, j, shows the amount of heat having entered the engine since the zero time at o. It is gotten as follows: by means of

the "steam supplied" and "length
 of cut-off" from the several
 tests I, II, III, & IV spoken of above,
 knowing the pressure and
 quality of the steam supplied,
 we can get the points $t, z, y, \text{ \& } x$,
 the heights of which represent
 the amount of heat having
 entered the engine since the
 time, σ , of head admission;
 from cut-off to the next ad-
 mission, no heat is entering
 the engine, hence the curve
 is a horizontal line from t to u ;
 the curve then rises from
 crank admission to crank cut-off
 in a manner similar to that
 of the head end. This curve
 shows, as was mentioned before, the
 amount of heat having entered

the the engine since the zero time, σ . It is shown in black on the diagram W.

The blue curve A, σ , B, C, D, E is to show the amount of heat leaving the engine in the exhaust. Knowing the weight and pressure of the steam leaving the engine during each exhaust and having measured its quality by the Carpenter Calorimeter, we can calculate the amount of heat rejected by the engine during each exhaust. Now from our "free expansion" idea in diagram Y the amount of heat $f-l$ has left the engine in the exhaust during the time from l to σ . Then let AL be equal to $f-l$ and

we have heat leaving the cylinder
 in the exhaust from A through
o to B. This curve passes
 through o since the horizontal
 through o is what was taken
 as the amount of heat actually
 in the engine at the point
 of head end admission. This
 curve from A to B is not
 necessarily a straight line
 but is merely taken as such
 since nothing definite is known
 about it. From compression
 at B to the cut release
 at C there is no heat leaving
 in the exhaust, hence curve
 from B to C is a horizontal
 line. We then have a repetition
 of affairs for the "head" exhaust,
 as in curve C, D, E.

Starting with the heat having entered the engine in a given time, if we subtract the heat rejected and the work done in that given time, we have left the amount of heat radiated to the outside air. Now we suppose that this radiation takes place uniformly with the time and thus we get a straight line σR for the radiation curve. The ordinates of this curve and also of the curve σW_k are laid off below the line OP simply for convenience.

From the indicator cards, we can measure the amount of work done, up to any given point. Then plotting

the curve showing work done (in B.T. U) we have, as shown in blue on the diagram W, the curve of work $\propto W_k$.

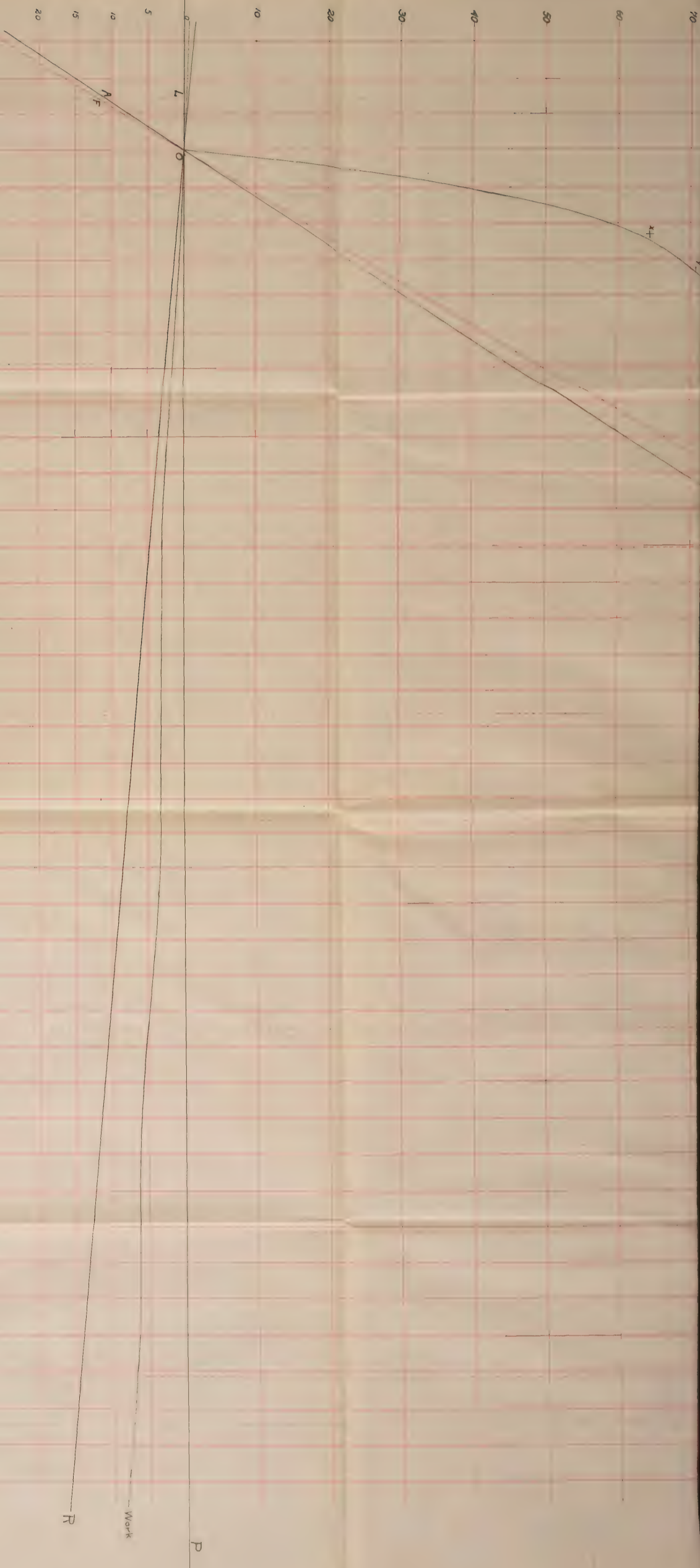
Then adding the ordinates of the curves $\propto R$ and $\propto W_k$ to those of the curve A, B, C, D, E, we have the "brown" curve F, O, G, H, K, J which shows the amount of heat disappearing i. e. accounted for, in the engine.

The diagrams W and Y are on page 79.

1" abscissa = .02 seconds.

1" ordinate = 10 B. T. U.

Heat(B.T.U.)



Crank Release.

Head Admission

Head Cut Off.

Crank Compression.

Head Release.

Crank Admission.

Crank Cut Off.

Head Compression.

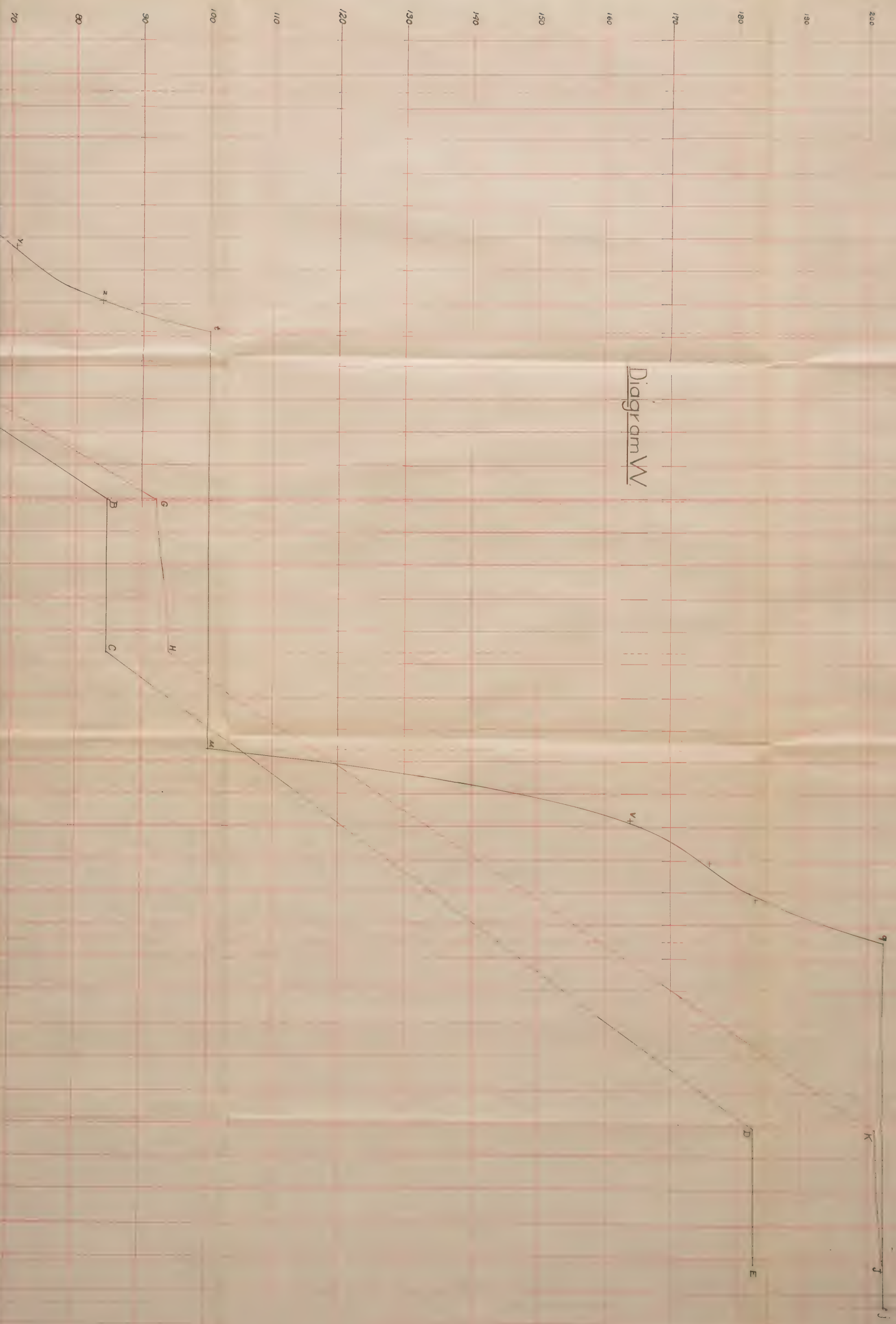
Crank Release.

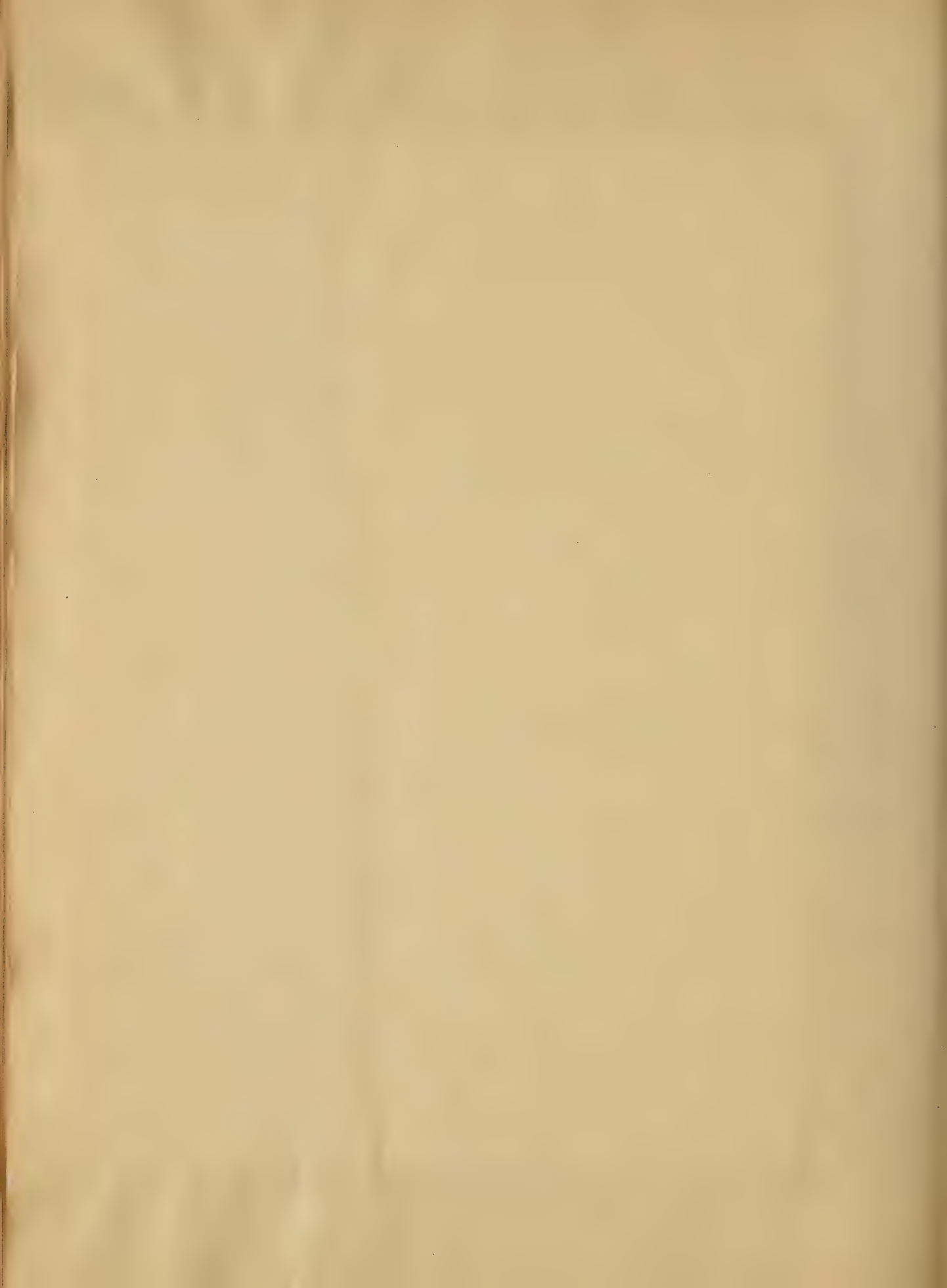
Head Admission.

Diagram Y

Time(Seconds)

Diagram VV





9- In drawing these time-heat curves and in making the calculations for this thesis some assumptions have been made, the fairness of which we now wish to discuss.

In getting the weight of steam in the cylinder during compression, it was assumed that at the point of compression, the quality of the steam in the cylinder was the same as that in the exhaust pipe. A little reflection will show that this is a very fair assumption and even if it were in error, a very small effect would be had on the final results.

The steam supplied to the engine per revolution was assumed to be divided between

the ends of the cylinder proportionally to the areas of the cards. The areas of the cards represent work done and the distribution of steam was thus made proportional to the work done. Perhaps a more fair assumption to make would have been to divide it proportionally to the time during which the admission valves were open, as from the diagram W or Y.

The assumption made in regard to the curve between admission and cut-off as explained on page 70 is as fair a one as could possibly be made under the circumstances.

The assumption of "free expansion" at release is, as will be

seen from the explanation on page 77 a quite fair one.

The assumption of straight lines in the upper part of diagram W is, as was said before, really groundless except for the fact that the points A, σ , & B happen to be in a straight line on the "blue" curve drawn.

That the radiation takes place uniformly with the time is without proof except for what we know of radiation in general.

There are doubtless many small errors in the curves drawn, due to graphical methods being employed.

10- From an inspection of the diagrams W and Y on page 79, we see that the heat ordinates of the work and radiation curves are very small as compared with those of the other curves. However, the steam consumption of this engine, as seen from the table on page 44, quite abnormal; and the ratio of the work curve ordinates to those of the other curves would ordinarily be somewhat materially larger.

Nevertheless we see from diagram Y that the total amount of heat held by the steam in both ends of the cylinder remains quite nearly constant, i. e. it is not subject to any

very great fluctuations. Since it is reasonable to suppose that the cylinder walls themselves are not subject to any very great and quick variations in temperature, does it not follow that the total amount of heat contained by the cylinder walls themselves is a quantity which remains nearly constant also. This would seem to lead us to believe that the total amount of heat held in the engine is quite nearly constant.

This state of affairs would also seem to be indicated from a combination of the "heat entering" and "heat disappearing" curves shown in diagram W.

That is, the amount of heat held by the engine varies to a not very large extent both above and below a mean value.

From a comparison of the curves of diagram W a suggestion comes that perhaps the heat entering one end of the cylinder during admission is immediately, by some manner or means, conducted or transferred to the steam of the other end and thus goes out in the exhaust. However one thing is very apparent, that is: the exhaust steam becomes heated up somehow. We have here shown that the quality of the steam at release is only about .4 to .5 while in the exhaust pipe it is somewhere

about .96. On expanding freely from the pressure at release down to the pressure of the exhaust, the value of x only rises slightly, such as from .4 to .45. Therefore, as was said before, it is very evident that a large amount of heat must be added to the exhaust steam at some time between release and the beginning of compression, i.e. the time when this exhaust steam has completely left the cylinder.

A fuller and more exhaustive study of this phenomenon along the line here indicated would, I feel sure, lead to a solution of

the problem under our
present consideration.





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